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ABSTRACT

A comprehensive model of a miniature-scale linear compressor for electronics cooling is presented. Linear compressors are appealing for refrigeration applications in electronics cooling. A small number of moving components translates to less theoretical frictional losses and the possibility that this technology could scale to smaller physical sizes better than conventional compressors. The model developed here incorporates all of the major components of the linear compressor including dynamics associated with the piston motion. The results of the compressor model were validated using experimental data from a prototype linear compressor. The prototype compressor has an overall displacement of approximately $3cm^3$, an average stroke of 0.6 cm. The prototype compressor was custom built for this work and utilizes custom parts with the exception of the mechanical springs and the linear motor. The model results showed good agreement when validated against the experimental results. The piston stroke is predicted within 1.3% MAE. The volumetric and overall isentropic efficiencies are predicted within 14.0% and 20.2%, MAE respectively.

1. INTRODUCTION

Miniature refrigeration systems offer distinct advantages for use in electronics cooling relative to other technologies. Only refrigeration offers the ability to cool components below the ambient temperature. The reliability and performance of semi-conductor devices is improved when operating at lower temperatures. Because vapor compression utilizes two-phase heat transfer in the evaporator, it is possible to maintain spatially uniform chip temperatures. The major concerns involving refrigeration systems are their cost and reliability, as well as miniaturization of the different components.

Trutassanawin et al. (2006) reviewed the available technologies for vapor compression systems in electronics cooling. Several prototype systems were discussed but there appeared to be no commercially viable solutions which are complete miniature systems. Chiriac and Chiriac (2008) concurred with this assessment in a recent review. Other investigations into refrigeration technology have led to the development of system prototypes (Mongia et al., 2006) and system-level models such as ones by Heydari (2002) and Possamai et al. (2008). However, it has been reported that the overall performance and size of these systems is still not at a level that is desired for desktop and portable electronic systems (Cremaschi et al., 2007). In particular the compressor is a critical component which can greatly affect the overall size and performance of the refrigeration system, as shown by Trutassanawin et al. (2006).

In electronics cooling applications, an atypical challenge for refrigeration systems is the relatively small temperature lift for cycle operation. The temperature lift is limited on the high-side by the environmental conditions and on the low-side by the desire to eliminate possible water vapor condensation from the ambient within the electronic package. Typically, this leads to a system with a small pressure ratio. The small pressure ratio leads to poor performance for currently available small-scale compressors, which are designed for refrigerator applications with a significantly larger pressure ratio. Some past studies using currently available technology have operated the compressors at a larger pressure ratio, and instead, developed solutions to handle moisture condensation by operating the evaporator and chip assembly in heavily insulated volumes (Heydari, 2002). The disadvantage of this option is a further increase in the cost of the cooling system.

A linear compressor is appealing for electronics cooling applications because it offers several potential advantages over traditional compressor technology. A linear compressor is a positive displacement compressor, similar to a reciprocating compressor. However, a linear compressor does not have a crank mechanism to drive the piston. Instead the piston is driven directly by a linear motor, as seen in Figure 1. The omission of the crank mechanism significantly reduces the frictional losses associated with conventional reciprocating compressors. It also allows oil-free operation which is a significant advantage with respect to the heat transfer performance of the condenser and evaporator in the system.

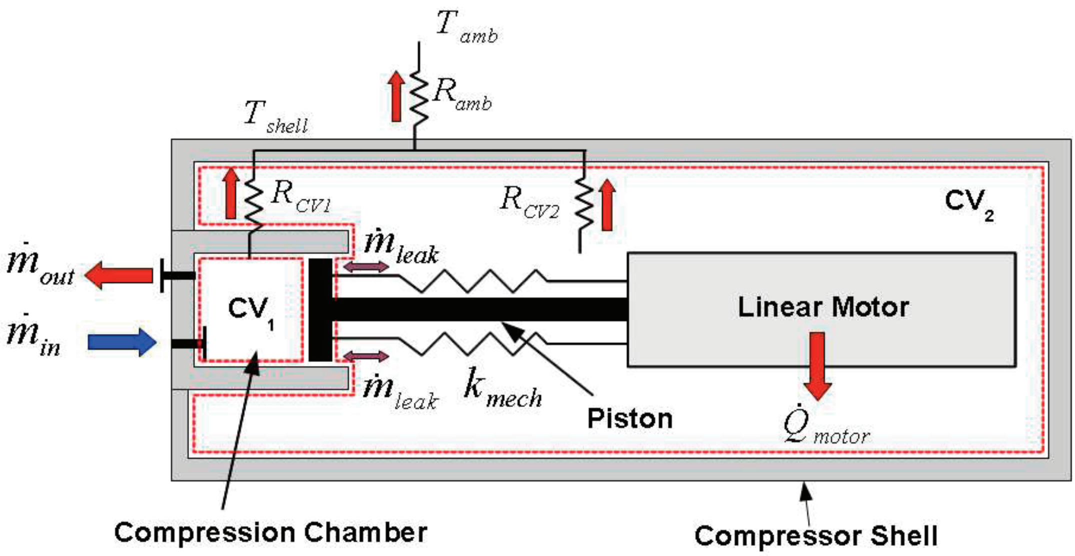

Figure 1: Representation of the compression process in a linear compressor.

In addition, a linear compressor is designed as a resonantly operated device. This means that mechanical springs are used to tune the device to operate at a resonant frequency. Operation at a resonant frequency allows a desirable reduction in size of the motor.

Early investigations of a linear compressor were conducted by Cadman and Cohen (1969a,b) for traditional refrigeration systems. Cadman and Cohen observed that the free piston operation of this device created some peculiar effects such as piston drift, which posed a challenge to modeling efforts and limited the practical application such devices. Pollak et al. (1979), investigated one-dimensional, nonlinear dynamics of the piston and electrical systems and confirmed such confounding effects.

Van der Walt and Unger (1994) assessed the state of the art in linear compressor technology. Unger and Novotny (2002) and Unger (1998) developed several linear compressor prototypes and performed a feasibility study which showed the potential for linear compressors in cryogenic and miniature systems. There have also been studies on the unique differences in the performance of linear compressors compared to conventional reciprocating compressors (Lee et al., 2001, 2004). Recently, Possamai et al. (2008) developed a prototype refrigeration system for a laptop cooling application which utilized a linear compressor. Another recent study on linear compressors investigated the sensitivity of the device to changes in operating frequency (Kim et al., 2009). This study utilized a single degree of freedom vibration model coupled to an electrodynamic motor model as was done by Pollak et al. (1979).

Past models of linear compressors have often employed linearizing assumptions or were restricted to a single degree of freedom in their analysis. Such analyses have either used a greatly simplified representation of device operation or have focused on specific components of the overall system. The present work develops a comprehensive compressor model which incorporates all of the major dynamics and nonlinearities as well as a second degree of freedom in the piston motion. In addition, the model predictions are validated using experimental data from a prototype linear compressor.

2. MODEL FORMULATION

The major components of the linear compressor model are described in this section. The comprehensive compressor model consists of a solution to two compression process equations. These equations require inputs from the five sub-models representing the valve flows, leakage flows, motor losses, heat transfer from the cylinder, and piston dynamics. Comprehensive compressor models of this nature have been developed for other compressor types (Chen et al., 2002a,b; Kim and Groll, 2007; Mathison et al., 2008).

2.1 Compression Process Equations

The compression process is modeled using mass and energy conservation over a control volume. In order to determine the state of the working fluid in the compression chamber at any point during the compression process, it is necessary to determine two independent fluid properties to fix its state. In this analysis the compressor is split into two control volumes as illustrated in Figure 1. The first is the compression chamber for the compressor, while the second consists of the remaining volume within the compressor. The compression process equations are solved for each control volume and coupled through the leakage, vibration, and heat transfer sub-models.

The fluid is assumed to be superheated throughout the entire compression process. The equation of state for the chosen working fluid, R-134a, is written in terms of temperature and density (Tillner-Roth and Baehr, 1994); the compression process equations are therefore cast in terms of these two variables.

By writing a mass and energy balance written over a general control volume and applying simplifying assumptions as seen in Bradshaw et al. (2010) equations for temperature and density can be written as follows:

$$m_{cv} C_v \frac{dT}{dt} + T\left(\frac{\partial P}{\partial T}\right)_v \left[\frac{dV}{dt} - \frac{dm_{cv}}{\rho dt}\right] + h_{cv}\frac{dm_{cv}}{dt} = \dot{Q} + \sum \dot{m}h_{in} - \sum \dot{m}h_{cv} \tag{1}$$

$$\frac{dm_{cv}}{dt} = \frac{d(\rho V)}{dt} = \rho\frac{dV}{dt} + V\frac{d\rho}{dt} \tag{2}$$

in which only temperature and density are the independent variables. The change in mass in the cylinder can then be written as:

$$\frac{dm_{cv}}{dt} = \frac{dm_{in}}{dt} + \frac{dm_{leak,in}}{dt} - \frac{dm_{out}}{dt} - \frac{dm_{leak,out}}{dt} \tag{3}$$

Equations (1) - (3) can now be solved simultaneously for the temperature and density in the two control volumes. Due to the nonlinear nature of these equations, a numerical solution approach is adopted. A number of inputs to this solution must still be determined from various sub-models as will be discussed below.

2.2 Valve Model

Reed valves are used in the present work as is typical for reciprocating compressors. These valves are essentially cantilevered beams which are displaced when a differential pressure is imposed across them. To ensure that pumping, as opposed to back flow, of the gas occurs two valves are used with limited ranges of operation. The valve body is constructed to only allow valve motion in the direction of desired flow for each valve this ensures that pumping occurs.

Valve analysis is split into two modes of operation depending on the location of the valve reed, similar to the analysis of Kim and Groll (2007). Early in its deflection the stagnation pressure driving the valve reed is assumed equal to the high side pressure. As the valve opens further, this assumption becomes less valid, as the movement of the valve is now dominated by the movement of fluid past the valve and the stagnation pressure becomes equal to the low-side pressure. These two modes of operation are referred to as the pressure-dominant and mass-flux-dominant modes, respectively. Models for each mode of operation are developed as 1-D, second order, dynamic systems. These models are presented in detail in Bradshaw et al. (2010).

2.3 Leakage Model

The leakage model only focuses on leakage past the piston. The only other leakage paths, those past the reed valves, are ignored since they are negligibly small compared to the leakage past the piston (Kim and Groll, 2007). The piston leakage is modeled as an incompressible Couette-Poiseuille flow driven by the pressure difference across the piston and the movement of the piston. This assumption is valid because the flow Mach number in the simulations is found to be between 0.1 and 0.3 at the representative conditions in the experiment.

2.4 Heat Transfer Model

The instantaneous heat transfer from each of the control volumes is calculated using the empirical approach of Fagotti and Prata (1998). By integrating these instantaneous heat transfer rates over the entire cycle, the total heat transfer from each of the two control volumes is calculated for use in the overall energy balance.

2.5 Vibration Model

A linear compressor is a free-piston device for which the stroke is not fixed by a crank mechanism but is instead determined by chosen geometry, the linear motor, and the mechanical springs used. The piston dynamics are modeled using a free body diagram of the piston assembly as shown in Figure 2.

Both the desired linear motion of the piston as well as its undesirable rotation due to eccentricity in the mechanical springs are considered. Any eccentricity causes a perpendicular reactionary load from the mechanical spring as it is compressed, since the spring cannot react to a load along its entire circumference. Thus, the effective load acts at a point offset from the center of mass of the piston. In a traditional compression spring as used in the prototype fabricated for this work, this point is assumed to be along the circumference of the spring where the edge of the spring contacts the piston assembly.

The linear motion of the piston as well as its rotation, are modeled as a two-degree of freedom vibration system. By assuming that the piston is a rigid body and the motions in the y and z directions as well as the rotation are small, the following equations describe the motion of the piston:

$$M\ddot{x}_p + c_{eff}\dot{x}_p + (k_{gas} + k_{mech})x_p = k_{mech}\varepsilon\theta + F_{drive} \tag{4}$$

$$J_{CG}\ddot{\theta} + k_{mech}\varepsilon^2\theta = k_{mech}\varepsilon x_p \tag{5}$$

In addition, it is assumed that the driving force and the reactionary forces from the compression chamber are applied directly through the center of mass of the piston. k_{gas} as well c_{eff} are non-linear coefficients in Equation (4). The calculation of these terms is shown in Bradshaw et. al (2010).

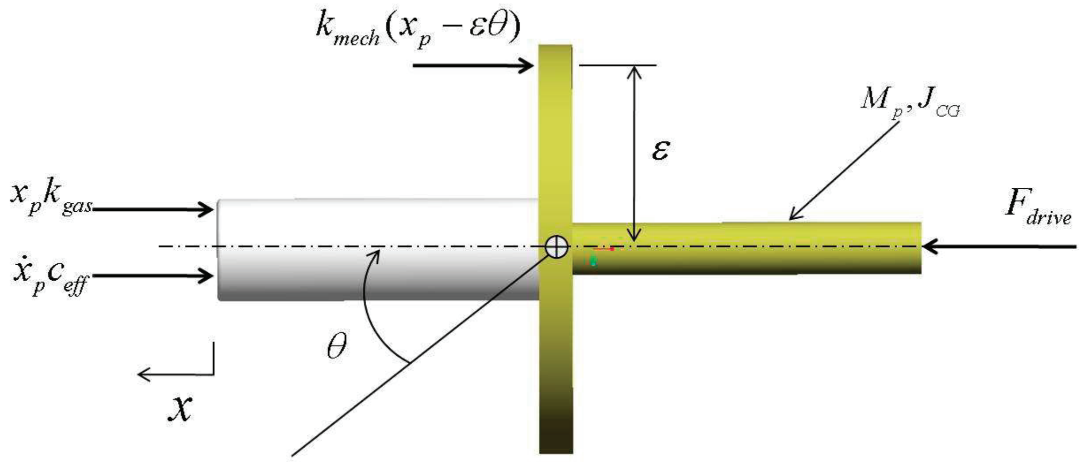

Figure 2: Free body diagram of linear compressor piston assembly.

2.6 Overall Energy Balance

To ensure that all of the compressor components satisfy an energy balance, a thermal network is constructed to account for heat transfer from the compression chamber (Chen et al., 2002a, Kim and Groll, 2007, Mathison et al., 2008). The energy balance for the compressor assumes that the heat transfer between the two control volumes is negligible and that the heat only flows to the compressor shell. A lumped-mass thermal network can then be constructed consisting of a single lumped mass to represent the compressor shell with two heat inputs and one heat output to the ambient. The heat output path is calculated using a forced convection correlation for flow over a cylinder to determine the thermal resistance between the compressor shell and the ambient (Hilpert, 1933). The thermal network elements are also shown in Figure 1. This network adds the following relation which is solved simultaneously with the compression process equations.

$$\frac{T_{shell} - T_{amb}}{R_{amb}} = \dot{Q}_{cv1} + \dot{Q}_{cv2} \tag{6}$$

2.7 Solution Approach

The model developed above for the linear compressor consists of two non-linear first order differential equations for the compression process and one non-linear equation from the energy balance. An explicit closed-form solution is not available, and the equations are instead solved numerically. The compression process is discretized into small time steps and each equation is solved using a fourth-order Runge-Kutta method.

The key input guesses to the model are the compressor inlet pressure and temperature, and the desired discharge pressure. For each time step, the model sequentially steps through each sub-model and calculates the solution inputs for the next time step. Once each sub-model has been called the overall compression process solver is called which solves Equations (1) to (3) and calculates the internal state in the compressor. When the model has iterated through an entire compression cycle Equation (6) is solved for T_{shell}. When the changes in temperature and density in each of the two control volumes as well as the change in T_{shell} is less than 0.01%, a converged solution is available.

3 EXPERIMENTS

No linear compressors are commercially available in the capacity and pressure ranges desired for electronics cooling. Therefore, a prototype compressor was custom-designed and built for the purpose of conducting experiments that could serve to validate the model developed in this work. The compressor was built using a moving-magnet type linear motor (H2W Tech). The housings, piston, and valve assembly were built in-house. A cross-sectional view of the prototype compressor is shown in Figure 3, with key dimensions given in Table 1.

Table 1: Linear compressor prototype parameters.

k_{mech}	f	A_p	g	η_{motor}	ε	$V_{cyl,max}$
N/m	-	cm^2	mm	-	cm	cm^3
23000	0.35	1.217	0.37	0.417	1.145	3.091

The prototype linear compressor was tested on a compressor load stand specifically built for testing miniature-scale compressors. The compressor load stand is based on a hot-gas bypass design; a schematic is shown in Figure 3. This type of load stand has been used successfully in previous studies to conduct performance tests compressors (Sathe et al., 2008; Hubacher et al., 2002).

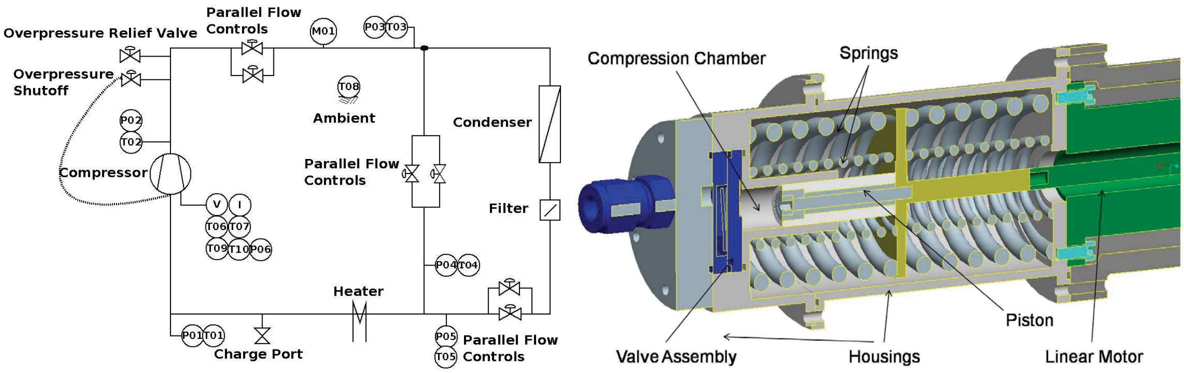

Figure 3: Schematic of experiment load stand and cross sectional view of prototype compressor.

The compressor was tested under a range of operating conditions, which simulate an electronics cooling application. At each specific set of compressor inlet conditions, the compressor was operated until a steady state condition was achieved, at which point data was collected. This process was then repeated for subsequent compressor inlet conditions. It is noted that the performance testing of a linear compressor is significantly different from that of conventional positive displacement compressors. During the performance testing of conventional compressors, the stroke and frequency of the piston are naturally held constant due to the kinematics of the device. However, a linear compressor requires that the frequency be adjusted to maintain operation at resonance. The power input and operating frequency are varied in these tests so as to maintain similar stroke values throughout the testing. The state points achieved in the experiments are summarized in Bradshaw et al. (2010).

4 VALIDATION OF MODEL PREDICTIONS

The experimental results are compared to the predictions from the model. The model performance is quite sensitive to several parameters: the leakage gap between the piston and cylinder, g, the eccentricity of the piston, ϵ, the dry friction coefficient, f, and the linear motor efficiency, η_{motor}. Since these parameters are difficult to measure, best estimates were used and adjusted within reasonable ranges to provide the best fit with the experimental data. These parameter values are summarized in Table 1.

The parameters in Table 1 plus the suction and discharge pressure, suction temperature, frequency, and input power with the same values as in the experiments were used as inputs to the model. Figure 4 shows the model predictions compared with the experimental results for piston stroke, mass flow rate, volumetric efficiency, and overall isentropic efficiency respectively. The model predicts the stroke of the linear compressor within 1.3% MAE. The predicted mass flow rates agree with the measured values to within 9.5% MAE. The predicted volumetric efficiency shows a similar behavior to the mass flow rate, predicting the experimental results to within 14.0% MAE. In the performance testing of conventional compressors the volumetric efficiency is simply a dimensionless mass flow rate. However, for a linear compressor, the volumetric efficiency depends not only on the mass flow rate but also the frequency. The piston stroke also varies slightly depending on the operating condition, which affects the volumetric efficiency. The overall isentropic efficiency is predicted by the model to within roughly 20.2%.

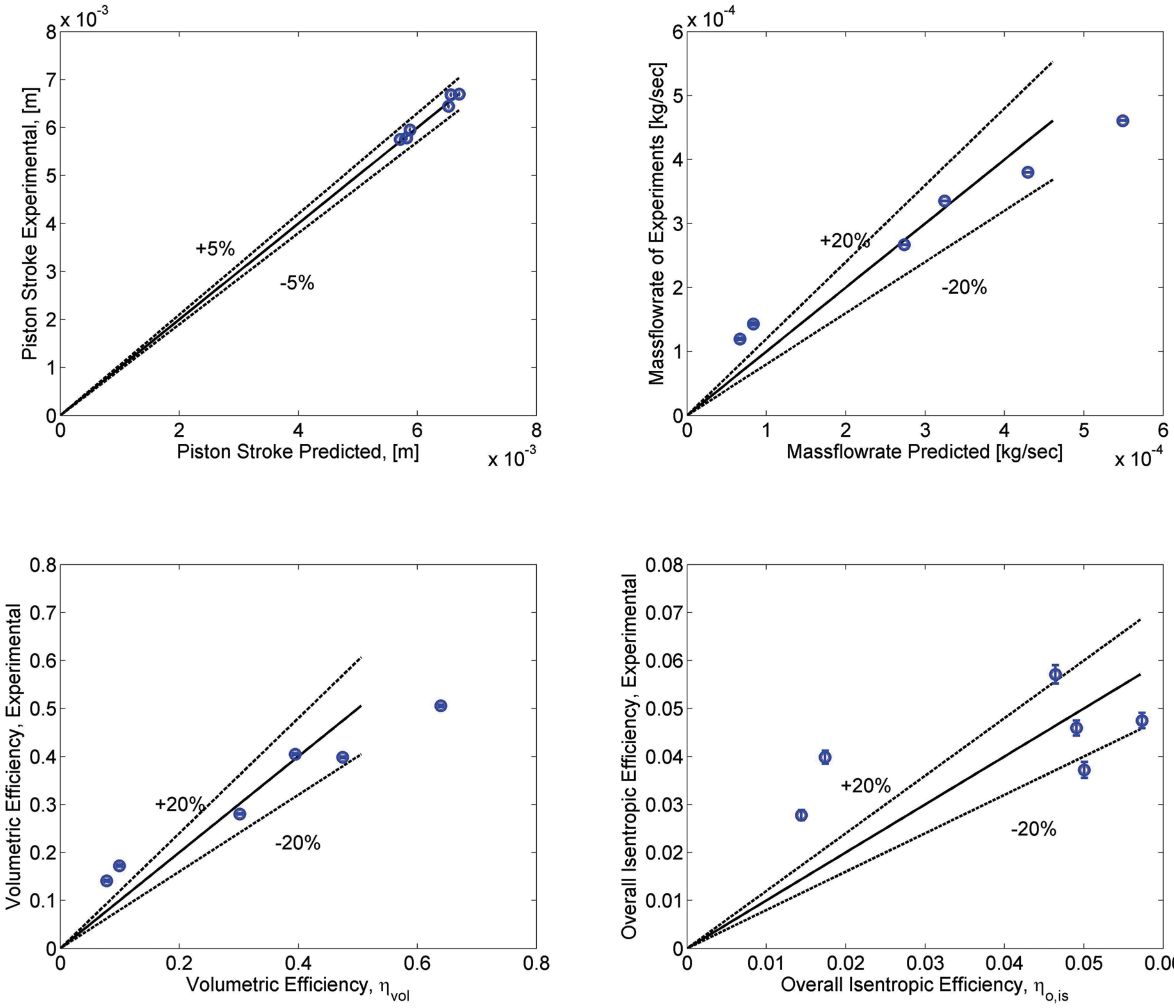

Figure 4 : Experimental over predicted values for stroke (top left), mass flow rate (top right), η_{vol} (bottom left), and $\eta_{o,is}$ (bottom right).

The model prediction of stroke, with good agreement, is evidence that the vibration sub-model captures the dynamics of the compressor piston well. The mass flow rates show higher discrepancy which suggests that either the valve or leakage models predict the physics to a lesser degree. In other compressor models these flow paths are modeled using semi-empirical models for high accuracy agreement to a particular compressor (Mathison et al., 2008, Kim and Groll, 2007, Navarro et al., 2007). The models presented here are based on first-principals in an effort to add flexibility to the model for use in parametric study. The larger discrepancy of the volumetric and overall isentropic efficiency can then be attributed to contributions from discrepancies of these sub-models.

5 CONCLUSIONS

A comprehensive model of a miniature-scale linear compressor for electronics cooling applications is developed. The model consists of a vibration sub-model that is particular to the operation of a linear compressor as well as other sub-models for valve flow, leakage past the piston, and heat transfer. The valve and leakage models are developed from first principals. The model is shown to predict the dynamic behavior of the piston well. Both the trends and quantitative values of the mass flow rate, volumetric, and overall isentropic efficiencies, respectively, are also predicted to within reasonable bounds.

In future work this model can be used in a series of parametric studies to determine an optimum design for electronics cooling. Starting with a desired operating condition the model can be used to optimize the compressor stroke, cylinder diameter and leakage gap. In addition, the model can be used to determine the correct mechanical spring rate and required motor power which are used to estimate overall package size.

NOMENCLATURE

A	area, m^2
C_v	specific heat at constant volume, J $kg^{-1}K^{-1}$
F_{drive}	driving force from linear motor, N
J_{CG}	rotational moment of intertia, kg m
M	moving mass, kg
P	pressure of gas, kPa
R_{amb}	thermal resistance, W K^{-1}
T	temperature of gas, K
V	volume of control volume, m^3
$\dot{Q}$	heat transfer, kW
$\dot{m}$	mass flow rate of gas, kg s-1
c_{eff}	effective damping of piston, N s m-1
f	dry friction coefficient, -
g	leakage gap width, m
h	specific enthalpy, kJ kg-1
k	stiffness or spring rate, N m^{-1}
t	time, sec
v	specific volume of gas, m3 kg-1
x	displacement, m

Greek Letter

ε	eccentricity of spring force, m
η	efficiency, -
ρ	density of gas, kg m^{-3}
θ	rotational angle of piston, rad

Subscripts

cv	control volume
in	into control volume
leak	leakage
mech	mechanical
motor	quantity from motor
o,is	overall isentropic
out	out of control volume
p	piston
shell	compressor shell
v	constant volume process
vol	volumetric

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